Computational Fluid Dynamics Application for Analysis of Centrifugal Compressor Stage Stator Part

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***Abstract*—The known information of satisfactory correlation of calculated and measured stator part performances is a foundation for the numerical investigation. Stator parts «Vane (vaneless) diffuser + crossover + return channel» of stages with different specific speed were designed in accordance with standard recommendations and investigated by CFD calculations. Flow structure demonstrated advantages and disadvantages of design. Flow separation in crossovers was eliminated by its shape modification for stages with diffusers relative width of diffusers. The stage with medium flow rate and low loading factor was designed with traditional and modified crossovers. Calculated efficiency performance becomes better. The information obtained is useful for design method better calibration.**

***Index Terms*—gas dynamic design, pipeline compressor, test results performance, centrifugal compressor, efficiency, mathematical model**

NOMENCLATURE

b - width of channel; c_2 - absolute velocity at an impeller exit; D_1 - blade inlet diameter; D_2 - impeller diameter; p - pressure; R - radius of curvature; T - temperature; α absolute velocity angle related to tangential direction; ζ - loss coefficient; η - efficiency; π - pressure ratio; ψ_T - loading factor; i - incidence angle; k - isentropic exponent; Φ - flow rate coefficient.

Subscripts: 2, 2-3, 3, 4, 5, 6, 0' - indices of control sections; des - design flow rate; p – pressure; inl – inlet; out – outlet; mean – mean; bl – blade; exter – external; opt – optimum; t - stagnation parameters.

Abbreviation: CFD - Computational Fluid Dynamics; CR - crossover; ST - stator part of a stage; RC - return channel; VD - vane diffuser; VLD - vaneless diffuser.

I. OBJECT AND AIM

Industrial centrifugal compressors consume a lot of energy. Their efficiency depends equally on impeller and stator part proper design. The cross-section of a model stage designed in accordance with popular recommendations [1] is presented in Fig. 1.

Typical stator part flow path consists of vaneless or vane diffuser, crossover and return channel. Flow path dimensions are presented in Fig. 2.

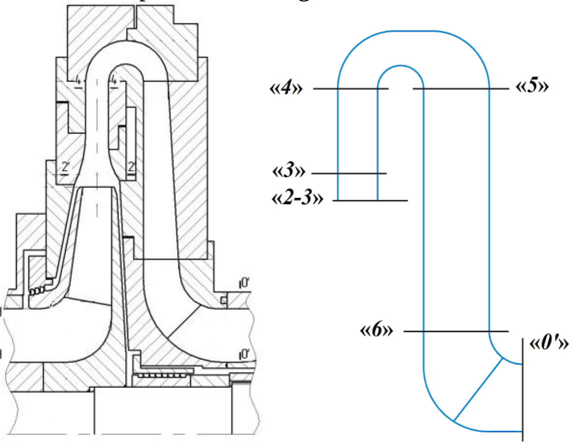

Figure 1. Typical industrial compressor stage ready to model test and stator elements borders.

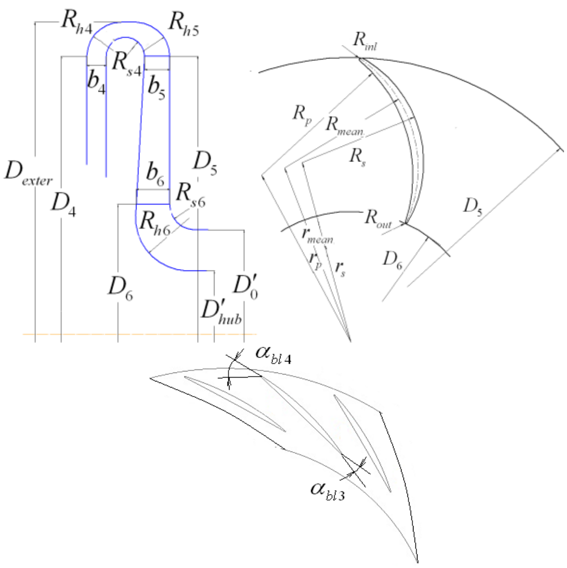

Figure 2. Typical industrial compressor stage part main dimensions.

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Configuration of all stage's elements is a combination of straight lines and arcs. Width of a diffuser related to an impeller diameter b_3 / D_2 must be is chosen in accordance with necessary flow angle at a design flow rate α_{3des}. Its value is about 250. Stator radial dimension is limited by a body size. Ratio D_3 / D_2 lies in range 1.4 – 1.7 as a rule. Inner bent radiuses are related to channel width R / b with proper recommendations, etc.

Several stages with vane and vaneless diffusers were designed in accordance with recommendations [1]. The authors executed CFD-calculations of gas dynamic performances of these stages and of several candidates. Alterations of crossover configuration and vane numbers were studied. Better efficiency and less loss coefficients have pointed on recommendations' alteration.

II. Calculation Method

Numerical calculations were performed using software packages ANSYS CFX 14 and NUMECA Fine Turbo. Calculation methodology, studied by the authors of [2], [3] was applied. To create a three-dimensional model programs BladeGen and AutoBlade were used. Grids were created by the program TurboGrid and AutoGrid. Computational domain consists of 700000-1500000 elements for different sectors. Grid of C-type is around blades and vanes. The rest computational domain is a grid of H-type.

NUMECA Fine Turbo grid consists of 870 000 for impeller's sector and 1250 000 for ST sector.

III. Calculated Data Representation

Polytrophic efficiency of a stator part based on calculated parameters of gas is:

$$\eta = \frac{\ln\left(p_{0'} / p_{2-3}\right)}{\frac{k}{k-1}\ln\left(T_{0'} / T_{2-3}\right)} \tag{1}$$

The important parameter of any diffuser and ST in particular is its velocity ratio:

$$\dot{c}_{ST} = \frac{c_{0'}}{c_{(2-3)}} \tag{2}$$

Loss coefficient that is followed from Bernoulli's equation:

$$\zeta = (1-\eta)\left[1-\left(\frac{c_{0'}}{c_{(2-3)}}\right)^2\right] \tag{3}$$

$$\zeta = \frac{p_{t\,inl} - p_{t\,out}}{0,5(p_{inl} + p_{out})} \cdot \frac{2}{c_{inl}^2} \tag{4}$$

recovery coefficient:

$$\xi = 1 - \left(\frac{c_{out}}{c_{inl}}\right)^2 - \zeta \tag{5}$$

If a velocity ratio is close to 1, a polytrophic efficiency is close to 0 while losses level is not too high. Parameters (2) and (3) or (4) in case of low Mach number are sufficient.

Coefficients listed above are presented as functions of an incidence angle in case of ST with vane diffuser:

$$\iota_3 = \alpha_{bl3} - \alpha_3 \tag{6}$$

Inlet flow angle α_{2-3} is an argument in case of ST with vaneless diffuser. To compare performances of ST of stages with different specific speed the authors of introduce ST specific speed coefficient:

$$K_{n\,str} = \sqrt{\Phi_{(2-3)} / \psi_T} \tag{7}$$

where:

$$\left(\Phi_{(2-3)} / \psi_T\right)_{des} = \left[\Phi / \left(\frac{\rho_{(2-3)}}{\rho^*_{inl}}\psi_T\right)\right]_{des} = \left(4\frac{b_{(2-3)}}{D_2} tg\alpha_{(2-3)}\right)_{opt} \tag{8}$$

IV. CFD Validation

The validation of CFD-calculations was made by the authors of [3]-[5].

Calculations' comparison with experiments presented in [4] demonstrates the next:

- loading factor at a design flow rate ψ_{Tdes} is overestimated on 6 – 12%,
- performance in a whole has tendency to shift to higher flow rate. Shift is more evident in case of 3D impellers,
- maximum efficiency of a stage is estimated quite well in spite of above mentioned.

Data shows at the same time that stator part performances are modeled less controversy.

Measured and calculated performances match well in all range of flow rate coefficient. Loss coefficient value of crossover + return channel matches satisfactory. Flow non-uniformity and unsteady character after an impeller is most. Evidently its modeling by CFD is not quite satisfactory. Calculated loss coefficient minimal value for the ST in a whole is about 25% less than measured.

Calculation by ANSYS and NUMECA also demonstrates not complete matching. The authors have designed a medium flow rate stage and executed CFD modeling of performances. Fig. 3 presents some results.

Here again good flow rate range matching takes place and impeller influence is evident. Minimal loss coefficient calculated by ANSYS and NUMECA is different visibly. Anyway, the authors' opinion is that stator part candidates can be compared by CFD-calculated performances. The more effective candidate is by CFD calculation the more effective it is in reality. Good calculated flow structure correspondence to experimentally proven flow character is presented below.

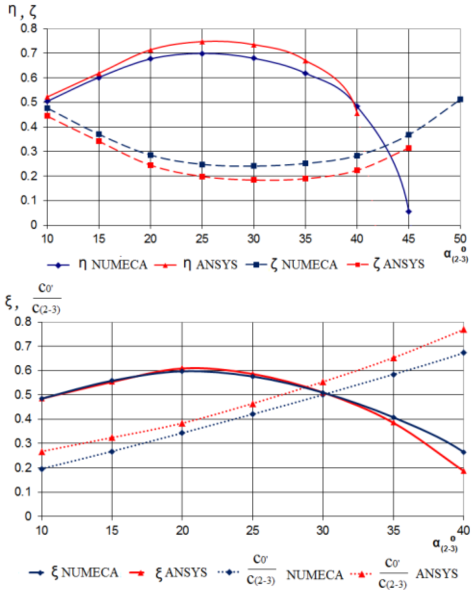


Figure 3. Stator part performances of a medium flow rate stage. Red – ANSYS, Blue – NUMECA.

V. Flow Structure

The authors have participated in design and calculation of the stage series presented in [6]. The stages with vane and vaneless diffusers were designed for a wide range of specific speed. Stator part performances are show in Fig. 4. Candidates ##1-5 are stages with vane diffusers. Candidate 1-VLD is a candidate #1 with vanes of VD eliminated. Candidates ##7 and 8 are stages with VLD and proper design. Here the authors are concentrated of flow structure in the candidates meaning checking of design recommendations.

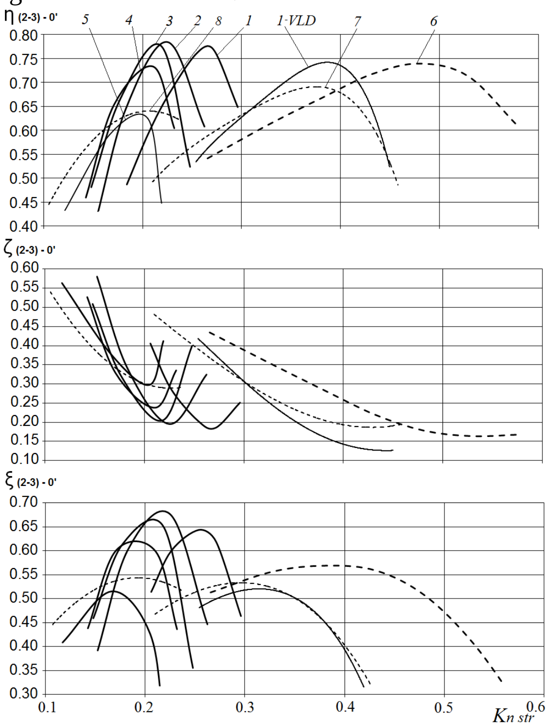


Figure 4. Performances of stators with VD and VLD designed for different specific speed.

Flow pattern near leading edges of the stage #1 at tree inlet angles is shown in Fig. 5 on the mean blade to blade surface. Angle α_{2-3} = 17° corresponds to the optimal regime. The optimal incidence angle is $i_{opt} = \alpha_{bl3} - \alpha_{2-3} = 21 - 17 = 4^0$. Streamlines show that at the angle $\alpha_{2-3} = 7^0$ (incidence angle 14^0) flow is fully separated. The local separation appears at the pressure side of blades when the angle $\alpha_{2-3} = 22^0$ (incidence angle -10).

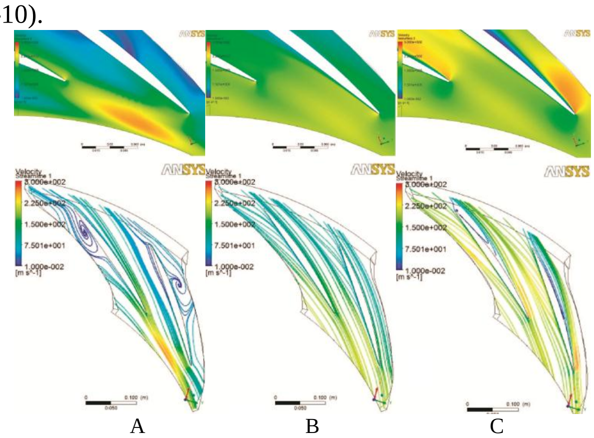


Figure 5. Velocity field near the leading edge and streamlines in the VD of the stage #1:A – α_{2-3} = 7°; B – α_{2-3} = 17°; C– α_{2-3} = 22°.

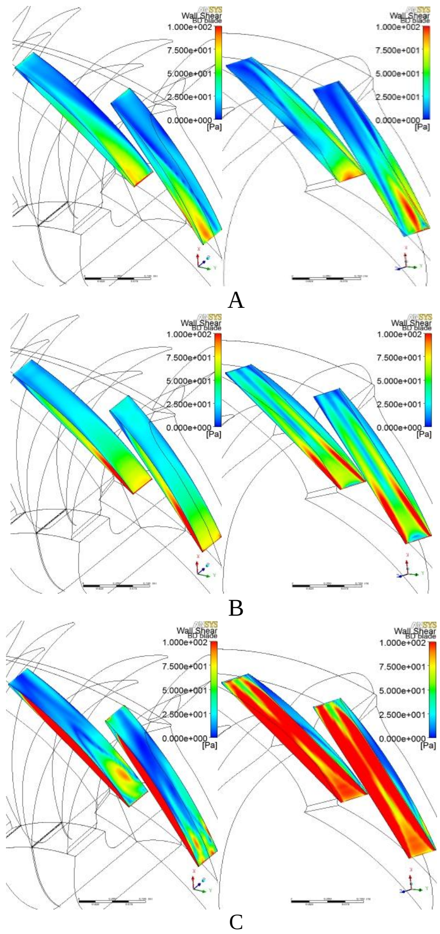
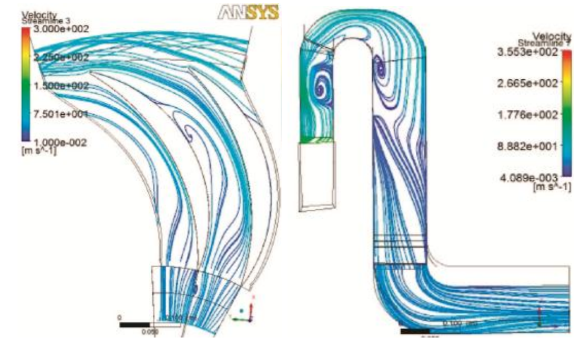


Figure 6. shear stresses on the VD blade surfaces (stage #1). above: pressure surfaces, below: suction surfaces. A – α_{2-3} = 7°; B – α_{2-3} = 17°; C– α_{2-3} = 22°

Fig. 6 shows shear stresses on the pressure and suction surfaces of the VD vanes. Flow separation corresponds to regions of low shear stress (blue). Red zones point on high level of friction losses.

Different level of flow separation takes place at all three inlet angles as streamlines in Fig. 7 demonstrate.



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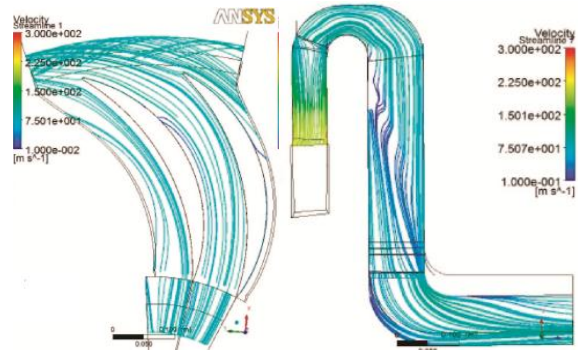

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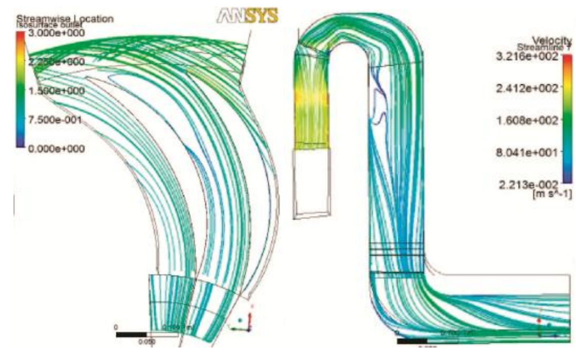

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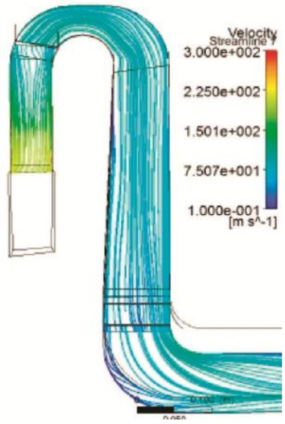
Figure 7. Streamlines in RC at three inlet angles. mean blade to blade surface and meridian plane. A – α_{2-3} = 7°; B – α_{2-3} = 17°; C– α_{2-3} = 22°

The flow deceleration in RC at the optimal inlet angle is $c_0 / c_4 \approx 0.9$. Evidently, it is necessary to avoid separation with better flow organization. Flow pattern is not quite favorable even at optimal mode. According to the date presented in [1] a separation is not expected in the crossover if the inner radius of curvature $R_s / b_4 \geq 0.6$ - as in the stages ## 1-5. Obviously, the used for design Universal modeling computer program of the 4th generation is not quite precise to predict flow behavior. It overestimates the importance of friction losses. The ratio b_6 / b_5 =1.0 was recommended by the program calculation. The flow deceleration in the crossover is too intensive as result. The crossover outlet/inlet ratio b_5 / b_4 is about 2.0 that leads to flow separation.

VI. Crossover Modifications

Meridian form of RC stages ##1 and 5 were modified for better organization of meridian flow. Modification was made by increasing the inner radius of curvature of crossovers by reducing the height of blades $\bar{b}_5$. Flow meridian structure in ST #1 before modification is shown in Fig. 7 above. In the modified stages flow separation takes no place at design flow rates. Fig. 8 demonstrates it.



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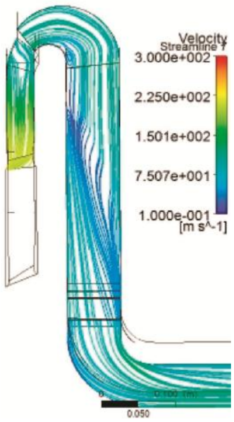

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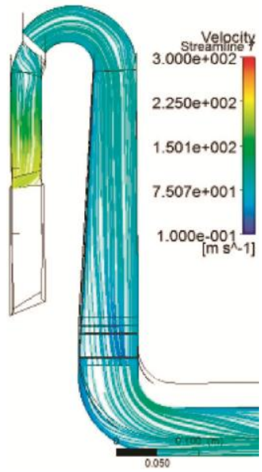

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Figure 8. Streamlines in meridian plane of RC at design flow rate. left – st #1 after. modification, middle- st #5 before modification, right – st #5 after modification

In modified ST meridian flow separation appears only at the extreme flow rates. Modification yielded positive results. Maximum efficiency of ST #1 has been increased by 1.6% by increasing of the inner radius of CR curvature and the reduction ratio b_5 / b_4 . Fig. 9 demonstrates how modification has improved gas dynamic performances of ST № 5. Modification have improved efficiency by 2,6%, despite the fact that in narrow ST friction losses predominate.

The authors of [6] have made a suggestion that further diminishing of a ratio b_5 / b_4 c may be a step in the right direction. Two candidates with extremely wide VLD b_3 / D_2 = 0.10 were compared (with b_5 / b_4 = 1, R_s / b_4 = 0.6, and b_5 / b_4 = 0.68, R_s / b_4 = 0.77).

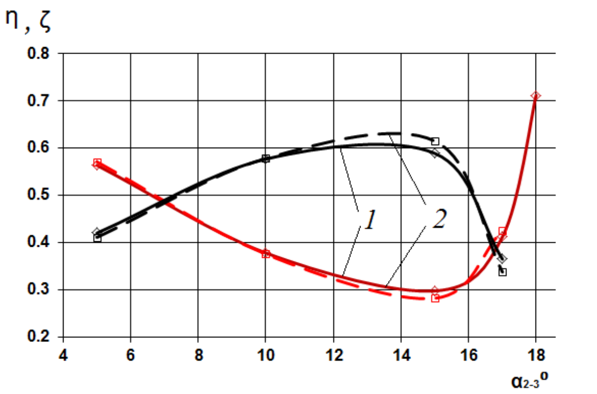

Figure 9. Efficiency and loss coefficient of ST #5 before (1) and after (2) modification

Optimal angle is α_{5opt} =31^0 and optimal incidence angle is i_{5opt} = +5^0 are values for the candidate before modification. The modified candidate has the same vane cascade. Due to smaller b_5 / b_4 ratio its optimal incidence is i_{5opt} = +9^0. The better flow organization and more effective operation of a stage with modified ST is expected.

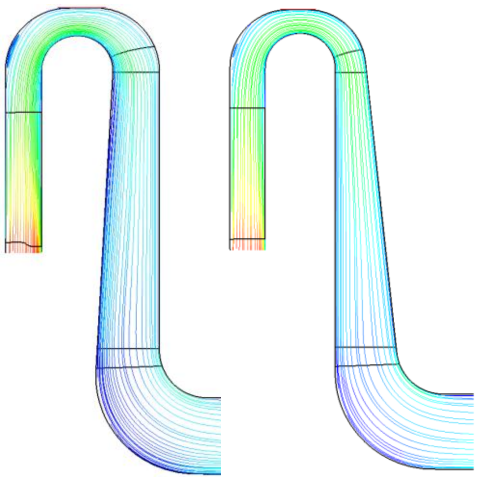

Figure 10. Meridian view of the stage with two candidates

The stage with design parameters Φ_{des} = 0.105, ψ_{Tdes} = 0.56 , was designed with two candidates of ST:

- with b_5 / b_4 = 1.3, R_s / b_4 = 1.77, close to standard recommendations;
- with b_5 / b_4 = 0.87, R_s / b_4 = 1.67.

Flow path meridian projection of both candidates is shown in Fig. 10 (left: b_5 / b_4 = 1.3, R_s / b_4 = 1.77, right: b_5 / b_4 = 0.87, R_s / b_4 = 1.67). The modification has lead to better efficiency of the stage. Performances of both candidates are presented in Fig. 11.

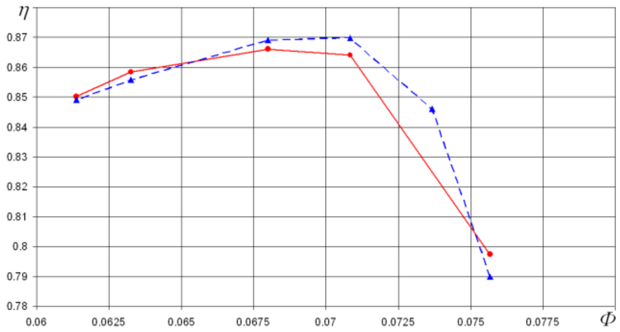

Figure 11. Gas dynamic performances of the stage with two candidates. Solid: b_5 / b_4 = 1.3, R_s / b_4 = 1.77, stroke: b_5 / b_4 = 0.87, R_s / b_4 = 1.67

VII. Crossover Modifications

The presented results of CFD calculations of centrifugal stages and their stators by software packages ANSYS CFX 14 and NUMECA Fine Turbo have demonstrated directions of possible improvements and some were realized. The optimum incidence angle for VD is 4^0. Optimal incidence of RC is angle is +5^0. Developed and tested recommendation on the confusor CR application.

There are many objects that will be studied by the authors by means of CFD-calculations and field-type optimal design programs that are at disposal.

VIII. Conclusion

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